

Generic design of ram pumps

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Abstract: The hydraulic ram pump was invented over 200 years ago but became obsolete with the general availability of mains water. It is used for pumping surface water and is powered by the potential energy of the supply. Ram pumps are particularly appropriate for use in remote regions of the developing world as their construction is basic and robust and they are inexpensive and easy to install and maintain. A rational design procedure applicable to any ram pump is described in this paper. The method is based on the three non-dimensional relationships that have been found to govern ram pump behaviour. The relationships express beat frequency, quantity delivered and source capacity in terms of the independent variables. By assigning upper and lower bounds to the critical velocity in the drive pipe, it is possible to derive three characteristic pump parameters from which the dependent variables may be determined directly. These parameters all take the same numerical values within a feasible design space. The design method has been confirmed using the results of existing experiments. A number of examples are given which illustrate the ease and rapidity of the procedure.

Keywords: hydraulic ram pumps, renewable resource, non-dimensional groups, characteristic parameters, general design

NOTATION

$a_{1,2,3}$	values of non-dimensional groups in equations (1) to (3)
A_d	cross-sectional area of drive pipe (m^2)
A_{disc}	cross-sectional area of valve disc (m^2)
c	velocity of sound (m/s)
C_c	cycle coefficient
C_d	discharge coefficient
C_Q	capacity coefficient
D	drive pipe diameter (m)
F_c	critical Froude number
g	acceleration due to gravity (9.81 m/s^2)
h	delivery head (m)
h_m	maximum delivery head (m)
H	supply head (m)
H_s	static head to close the impulse valve (m)
L	length of drive pipe (m)
L_{max}	maximum length of drive pipe (m)
m	valve loss coefficient
M_v	mass of impulse valve assembly (kg)
n	beat frequency (Hz)
q	quantity delivered (m^3/s)
Q	capacity of source (m^3/s)
Q_c	drive pipe flowrate at impulse valve closure (m^3/s)
Q_w	quantity wasted (m^3/s)

u_c	velocity to close impulse valve (m/s)
u_o	steady state velocity (m/s)
$X_{1,2,3}$	characteristic pump parameters (s or m/s)
ρ	density of working fluid (1000 kg/m^3 for water)
$\varphi_{1,2}$	functions of velocity ratio and head ratio

1 INTRODUCTION

The ram pump has been in use for over 200 years for pumping surface water to considerable heights (100–120 m) in regions lacking public utilities. The pump has only two moving parts, it is relatively inexpensive and, since it uses a renewable energy resource (the potential energy of the supply), it is environmentally friendly. A description of the mode of operation of a ram pump and a literature review have been given in a previous paper (1). The traditional method of ram pump design is unsatisfactory since it relies on rules-of-thumb and arbitrary assumptions for the independent parameters. The general approach described here has a rational basis and is corroborated by an extensive body of empirical evidence.

Figure 1 shows the main features of a ram pump system and identifies some of the parameters used here. Water is typically supplied from a dam (a) whose surface is at a height H above the pump unit. The supply Q is led to the pump through a drive pipe (b) of length L and diameter D . The pump has an impulse or waste valve (c) and a delivery valve inside an air chamber (d). High-pressure water flows

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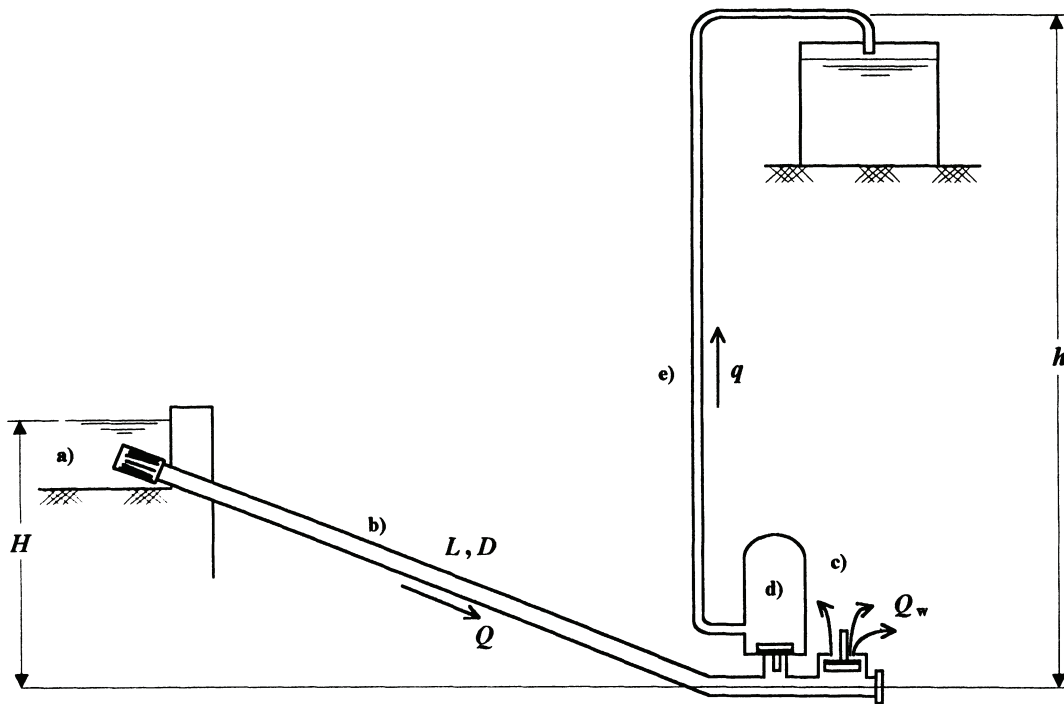


Fig. 1 Layout of a ram pump system

at a rate q up the delivery pipe (e) to a receiving tank at height h above the pump unit.

The author has shown (2) that the behaviour of a ram pump is governed by three equations involving non-dimensional groups. These equations could be used directly for design if the critical velocity u_c of the water in the drive pipe (the velocity that causes closure of the impulse valve) is known. The critical velocity is a complex function of valve geometry and mass characteristic of an individual pump and, since a designer is unlikely to be aware of the particular relationship for a chosen pump, it is an advantage to have a design procedure in which acceptable limits for u_c are implied. In the approach described here, the only impulse valve parameter needed is the static head H_s required to keep the valve closed, and this is easily determined from knowledge of the valve mass and the cross-sectional area of the valve disc if the valve is of the conventional poppet type.

2 THEORY

The non-dimensional equations referred to above were originally derived by consideration of ideal pump performance under zero recoil conditions. For a real pump, however, operating at any delivery head, they may be expressed as

$$\frac{nLu_c}{2gH} = C_c\varphi_1 = 0.29 \pm 0.05 \quad (1)$$

$$\frac{qh}{Q_cH} = C_d\varphi_2 = 0.27 \pm 0.05 \quad (2)$$

and

$$\frac{Q}{Q_c} = C_Q\varphi_2 = 0.5 \pm 0.1 \quad (3)$$

where n is the beat frequency (Hz), $Q_c (= A_d u_c)$ is the quantity flowing in the drive pipe at the instant of impulse valve closure (m^3/s), A_d is the cross-sectional area of the drive pipe, Q is the average flow in the drive pipe (m^3/s), g is the acceleration due to gravity (9.81 m/s^2) and φ_1 and φ_2 are functions of u_c , u_o (the steady state velocity) and the ratio h/H (2). The coefficients C_c and C_d (< 1) and C_Q (> 1) are included to reflect the less than ideal performance of a real ram pump. In previous work, C_Q has been taken as unity.

The numerical values assigned to the non-dimensional groups in equations (1) to (3) were obtained from the study of a large number of experimental results (2). Comparing these values with averages for the non-dimensional groups computed using a program described previously (1) suggests the values $C_c=0.8$, $C_d=0.75$ and $C_Q=1.1$ for the ranges of parameters in Table 1; the constraints $h \leq 0.6h_m$, $H < 0.1h_m$, $u_c < 0.8u_o$ and $h > 3H$ have been justified in earlier work (1, 2).

It has been shown (1) that a ram pump operates best at delivery heads $h \leq 0.6h_m$, where h_m is the maximum delivery head of which the pump is capable. Since $h_m = cu_c/g$ (Joukowski's law) and c (m/s) is the acoustic velocity of the water in the drive pipe, a lower bound for u_c is given by $u_c = hg/(0.6c)$. In the same paper, a relationship between u_c and H_s was presented as $u_c = \sqrt{(2gH_s/m)}$, where m is the impulse valve loss coefficient. Since m must be greater than unity, an upper bound for u_c is obtained as $u_c = \sqrt{(2gH_s)}$.

Table 1 Range of parameters used to compute coefficients in equations (1) to (3). Galvanized iron drive pipe ($c = 1000$ m/s, coefficient of roughness $k_s = 0.15$ mm)

$0.5 \leq u_c \leq 2.8$ m/s
$0.2 \leq H_s \leq 0.8$ m
$5 \leq L \leq 35$ m
$3 \leq H \leq 8$ m
$30 \leq n \leq 80$ min ⁻¹
$25 \leq D \leq 250$ mm
$h \leq 0.6 h_m$
$H < 0.1 h_m$
$u_c < 0.8 u_o$
$h > 3H$

The static head H_s is determined from $H_s = M_v / (\rho A_{\text{disc}})$ (on the assumption that the impulse valve is of the poppet type), where ρ is the density of water (1000 kg/m³), M_v is the mass of the moving parts of the valve and A_{disc} is the cross-sectional area of the valve disc.

Substituting the lower bound for u_c in equations (1) to (3), the upper bounds are found for three characteristic pump parameters as follows: $nLh/H = 1.2ca_1$, $HD^2/q = 2.4c/(\pi ga_2)$ and $hD^2/Q = 2.4c/(\pi ga_3)$ where, for convenience, $a_1 = C_c \phi_1$, $a_2 = C_d \phi_1$ and $a_3 = C_Q \phi_2$. Since $a_3 \approx 2a_2$, the latter condition may be written as $2hD^2/Q = 2.4c/(\pi ga_2)$. Substituting the upper bound for u_c , the lower bounds are obtained for the parameters as follows: $nLh/H = a_1 h \sqrt{(2g/H_s)}$ and HD^2/q and $2hD^2/Q = (4h/\pi a_2) \sqrt{(2gH_s)}$. For the numerical values of the upper and lower bounds to be identical for each of the three parameters, it is sufficient that the numerical value of the product $a_1 a_2$ is $2/(\pi g)$. Thus, if it is assumed that $a_1 = a_2$ (which is approximately the case), then $a_1 = a_2 = \sqrt{(2/\pi g)} = 0.255$. Hence, in the appropriate units, the common upper bound is $1.2c \sqrt{(2/\pi g)} = 0.306c$ and the common lower bound is $2h/\sqrt{(\pi H_s)} = 1.13h/\sqrt{H_s}$.

The three characteristic parameters are dimensional and may be represented collectively by X_n where $X_1 = nLh/H$ (m/s), $X_2 = HD^2/q$ (s) and $X_3 = 2hD^2/Q$ (s). The computer program was used to perform a large number of simulated tests (about 350 000) to evaluate real values of the three components of X_n (incorporating the coefficients C_c , C_d and C_Q) for various values of $h/\sqrt{H_s}$ over the range of parameters in Table 1. It was found that all results were contained within the quadrilateral OABC shown in Fig. 2a. The line OC of slope 1.13 (s/m^{1/2} or m^{1/2}/s) corresponds to the lower bound for X_n discussed above and the horizontal line AB ($X_n = 385$ s or m/s) is an upper bound. The value adopted for the acoustic velocity c is 1000 m/s to allow for the effect of entrained air in the drive pipe (1), but the magnitude of the upper bound in Fig. 2a is greater than the calculated value of 306 as a consequence of the simplifying assumptions and because average, rather than extreme, values of a_1 and a_2 were used. The left-hand boundary of the feasible design space was found to be represented by a line OB of slope 6.2 (s/m^{1/2} or m^{1/2}/s) and the right-hand boundary by a vertical line BC for which $h/\sqrt{H_s} = 192$ m^{1/2}. By eliminating u_c from equations (1) to

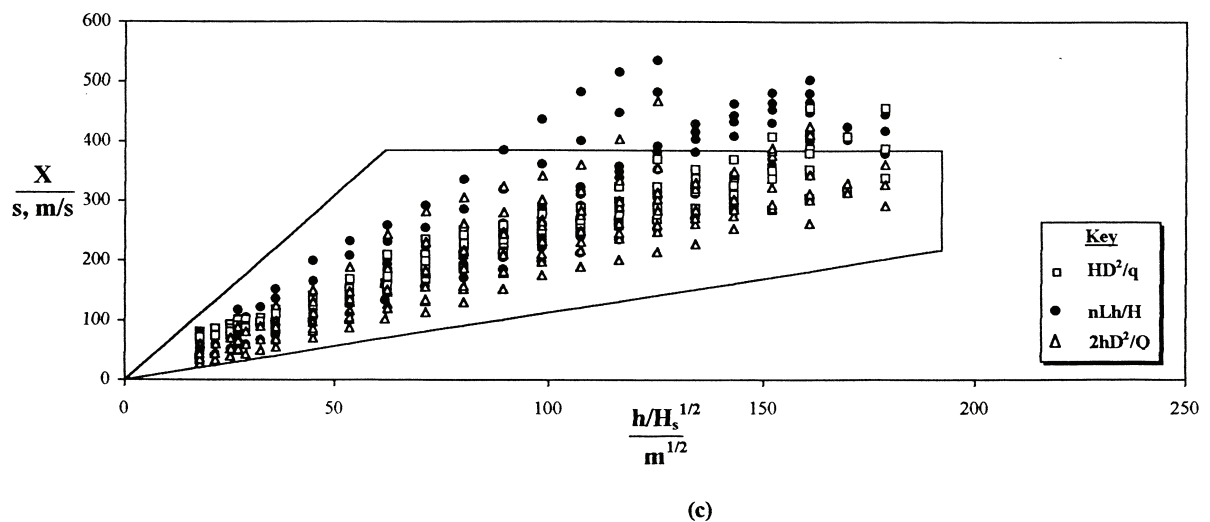
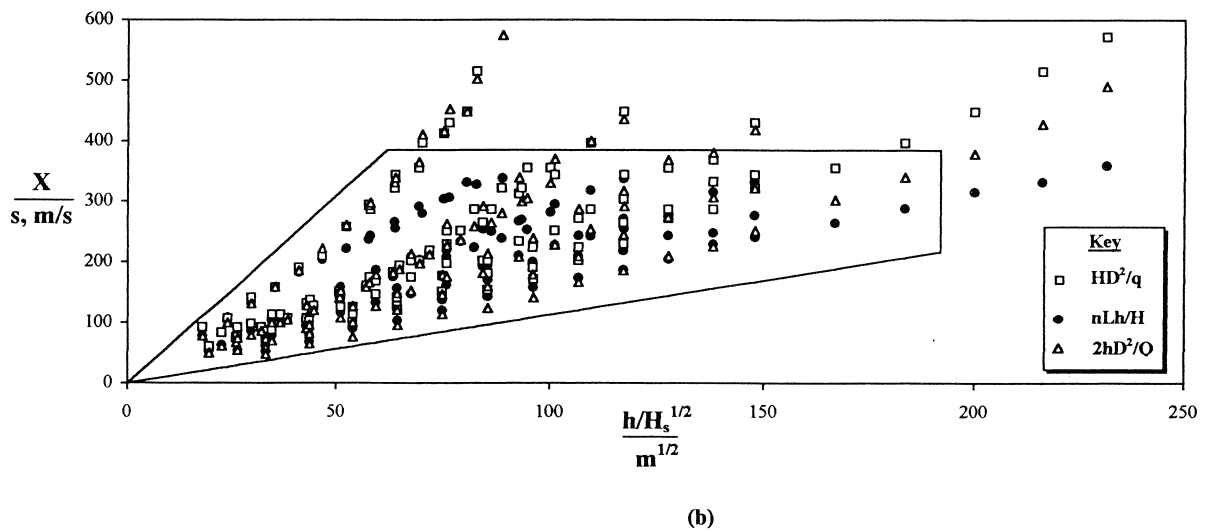
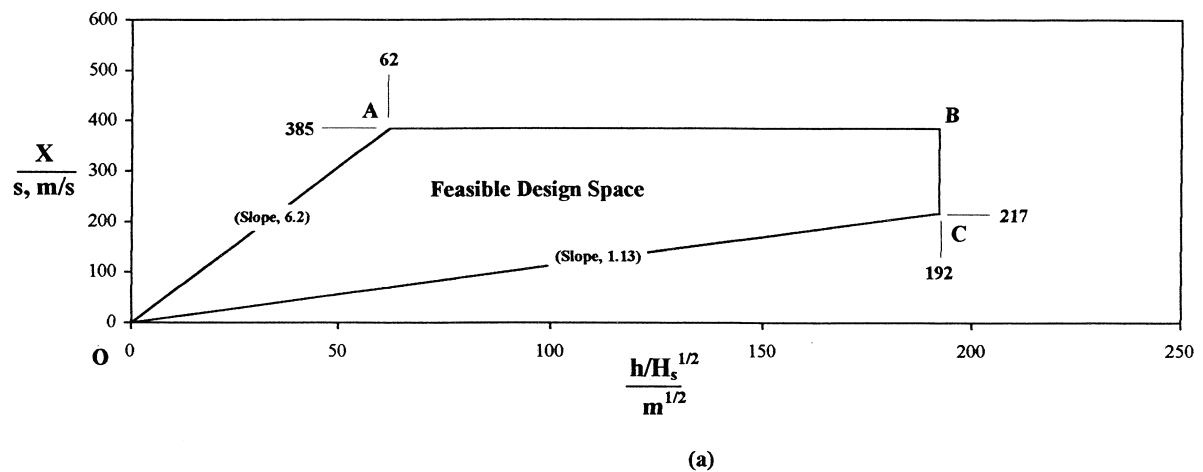
(3) it can be shown that numerical values of the three characteristic parameters are equal ($X_1 = X_2 = X_3$); thus any point in OABC represents a viable design for a particular pump with a given stroke and impulse valve mass. It is not necessary to check that the drive pipe length is less than the permissible upper bound L_{max} (2) since this condition is automatically satisfied if $u_c < 0.8u_o$. Also, from equations (1) to (3), the critical velocity may be estimated from $u_c = (2ga_1)(h/X_1) = (4/\pi a_2)(h/X_2) = (4/\pi a_2)(h/X_3)$ m/s. Inserting appropriate units gives $u_c = 5h/X_n$ m/s in which h and X_n take their numerical values. It follows that any straight line through the origin in the $X_n : h/\sqrt{H_s}$ plane has a slope of numerical value $(2/F_c) \sqrt{(2/\pi)} = 1.6/F_c$, where F_c is the critical Froude number given by $u_c / \sqrt{(gH_s)}$. The feasible design space is therefore bounded by values of F_c from 0.257 (OA) to $\sqrt{2}$ (OC).

3 EXPERIMENTAL VERIFICATION

Figures 2b to f show existing experimental results on the $X_n : h/\sqrt{H_s}$ diagram; only those results for which $u_c < 0.8u_o$, $n < 80$ min⁻¹, $h > 3H$ and $h < 0.6h_m$ have been plotted. Krol's results (3) from tests on a 51 mm pump appear in Fig. 2b. The valve strokes varied from 1.6 to 9.5 mm and the impulse valve mass was adjusted so that H_s took values between 0.066 and 0.543 m. The drive pipe length and supply head were 18.52 and 3.96 m respectively. The author's tests on a 53 mm Wilcox ram pump (1, 4) are shown in Fig. 2c. The supply head was 2.76 m for drive pipe lengths of 11.8, 15.1 and 18.3 m and 3.25 m for a drive pipe length of 15.1 m. Impulse valve strokes were from 6 to 16 mm in 2 mm steps and H_s was 0.33 m for all tests.

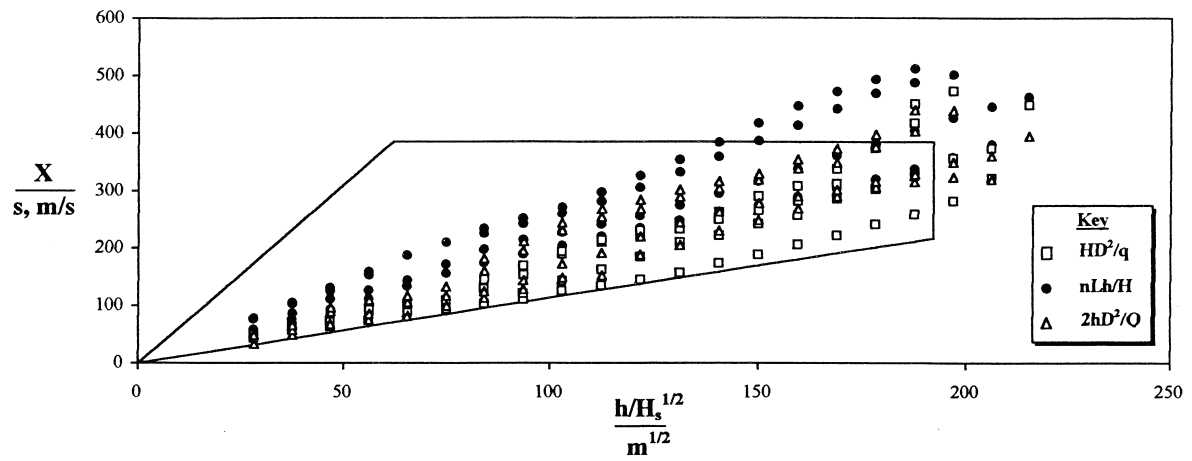
Further tests by the author on a 76 mm Platypus pump (5) are shown in Fig. 2d. The supply head was 3.44 m and the drive pipe length was 13.6 m. Valve strokes were 14, 18, 20, 22 and 26 mm and H_s was 0.3 m for all tests. Tests on a 26 mm pump of the Gould type were performed by O'Brien and Gosline (6); results are shown in Fig. 2e. The supply head was 3.54 m and the drive pipe length was 15.1 m. There were three valve strokes, 5.8, 10.9 and 18.5 mm, and H_s was 0.26 m. Finally, Fig. 2f shows previously unpublished results from tests carried out by the author on a 76 mm Billabong pump (which is also of the Gould type). The supply head was 3.28 m, the drive pipe length was 13.95 m, a single valve stroke of 20 mm was used and H_s was 0.5 m.

Tests on a wide selection of pumps have been carried out by Tacke (7). Drive pipe lengths were approximately 12 m and supply heads were 1 to 1.35 m, 2 m and 3 m. Unfortunately, details of the impulse valve mass and stroke were not reported and thus it is not possible to analyse the results in depth. Four pumps, however (two Blake Hydrams with $D = 38$ and 63 mm and two Vulcan rams with $D = 25$ and 51 mm), are of particular interest since their impulse valves consisted of a conical perforated seating fitted with a rubber disc, quite different from the poppet-type valves used in

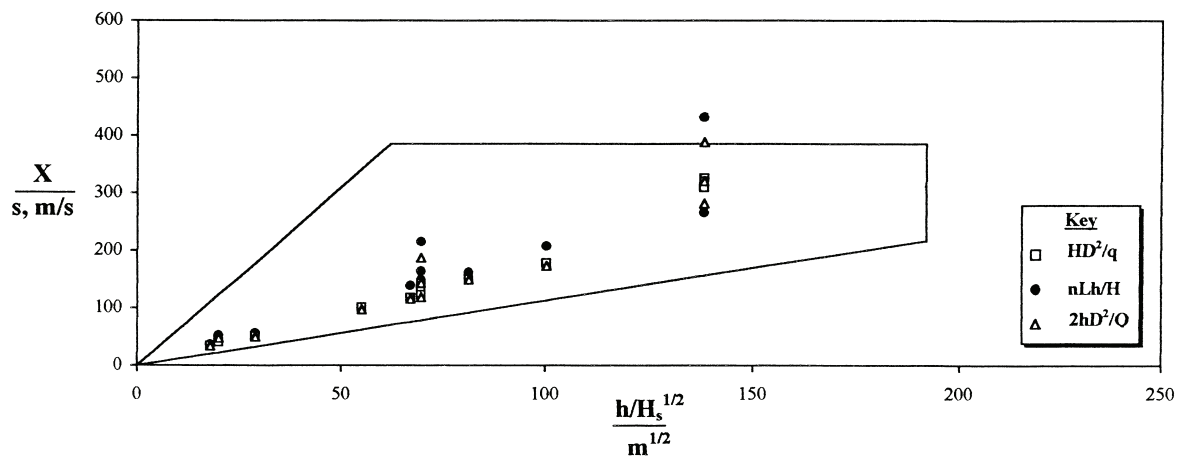


the tests referred to above. Ratios of the characteristic parameters (X_1 , X_2 and X_3) for all four pumps at supply heads of 2 and 3 m with $h < 0.6h_m$ and $h/H > 3$ varied between 0.8 and 1.5 with an average of 1.2. This is close

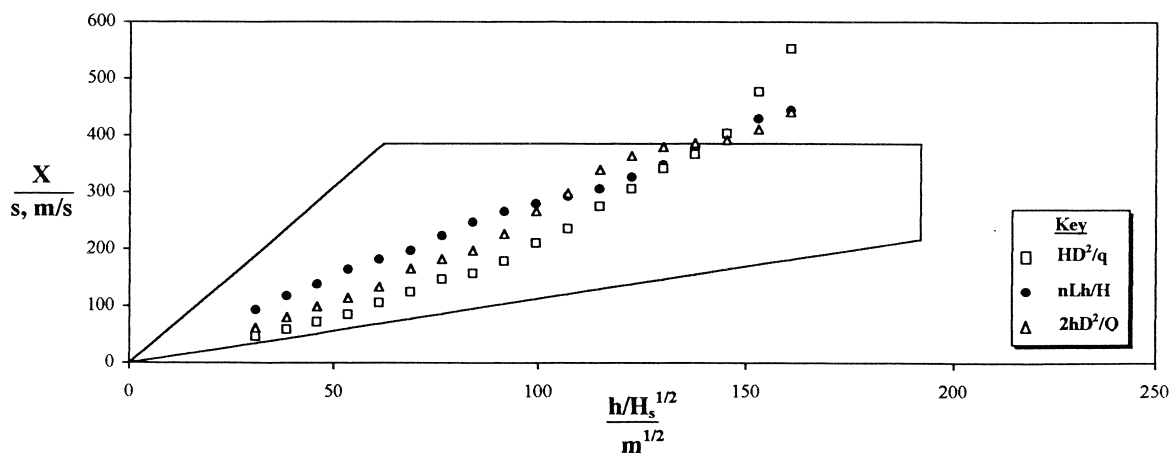
to the expected value of unity and suggests that the proposed approach is likely to be applicable to all ram pumps regardless of the design of their impulse valves. Values of the static head H_s for rubber disc valves cannot



(d)



(e)



(f)

Fig. 2 (a) The feasible design space; (b) Krol's results; (c) Wilcox ram results; (d) Platypus ram results; (e) O'Brien and Gosline's results; (f) Billabong ram results

Table 2 Various solutions for the first design example ($q = 8$ l/min, $h = 41$ m) with $10 < L < 30$ m and $H < 6$ m

X_n (s, m/s)	HD^2 (m ³)	D (mm)	H (m)	Q (l/min)	u_c (m/s)	nL (m/s)	n (min ⁻¹)	L (m)
100	0.013	51	5.13	128	2.05	12.50	30	25.0
		63	3.36	195		8.19	55	13.6
		76	2.31	284		5.63	30	16.4
150	0.020	63	5.04	130	1.37	18.44	30	11.3
		76	3.46	189		12.67	55	20.1
		102	1.92	341		7.03	80	13.8
		127	1.65	397		8.07	30	25.3
200	0.027	76	4.62	142	1.03	22.52	55	13.8
		102	2.56	256		12.50	80	14.1
		127	1.65	397		8.07	30	24.6
		127	1.65	397		8.07	55	16.9
250	0.033	76	5.77	114	0.82	35.19	30	25.0
		102	3.20	205		19.54	55	13.6
		127	2.07	317		12.60	80	16.1
		127	2.07	317		12.60	55	26.4
300	0.040	102	3.84	171	0.68	28.13	80	21.1
		127	2.48	265		18.15	55	19.8
		127	2.48	265		18.15	80	13.6
350	0.047	102	4.49	146	0.59	38.29	80	28.7
		127	2.89	227		24.70	55	26.9
		127	2.89	227		24.70	80	18.5

be calculated conveniently and are probably best determined by experiment.

Mead (8) reports data from three ram pumps in service with $D = 305$ mm and X_n in the range 64–130. Although these pumps were operated at h/H less than 3, average values for the ratios of the characteristic parameters were between 0.98 and 1.12. Fyfe (9) gives details of installed pumps with $D = 51$ to 127 mm. Values of X_n were between 38 and 363 and the average characteristic parameter ratio was 1.0. Anderson's tests on a 102 mm pump (10) also had the same average ratio.

4 DESIGN

The conditions for a feasible ram pump design may be summarized by reference to the diagram in Fig. 2a as follows:

For $h/\sqrt{H_s} < 62$ m^{1/2}:

$$\frac{1.13h}{\sqrt{H_s}} < X_n < \frac{6.2h}{\sqrt{H_s}} \quad (\text{s or m/s})$$

For $62 < h/\sqrt{H_s} < 192$ m^{1/2}:

$$\frac{1.13h}{\sqrt{H_s}} < X_n < 385 \quad (\text{s or m/s})$$

A lower bound for L has been recommended as 5 m (1) and experience has shown that preferred beat frequencies are between 30 and 80 min⁻¹.

Two examples to be examined in detail will serve to demonstrate how the design process works. In the first example, 8 l/min is to be delivered to a height of 20 m through a 32 mm nominal diameter polyethylene pipe of length 5 km. The friction head loss is estimated as 21 m and the total delivery head to be provided by the pump is therefore 41 m. Assume H_s to be 0.275 m; thus $h/\sqrt{H_s} = 78.2$ m^{1/2} and, since $62 < h/\sqrt{H_s} < 192$ m^{1/2}, $88.3 < X_n < 385$ s or m/s. Choosing $X_n = X_2 = 250$ s, for example, gives $HD^2 = 0.033$ m³ since the required delivery flow is 0.133×10^{-3} m³/s. If D is selected as 0.102 m, $H = 3.2$ m, which is acceptable. Now $X_1 = nLh/H = 250$ m/s; thus $nL = 19.5$ m/s and, selecting a frequency of $n = 0.8$ Hz (48 min⁻¹), $L = 24.4$ m. Finally, $X_3 = 2hD^2/Q = 250$ s, from which $Q = 3.4 \times 10^{-3}$ m³/s (205 l/min). The required beat frequency is set on installation by a suitable choice of impulse valve stroke. If necessary, the valve mass may also be adjusted since this is not critical. A fairly wide range of choices for H_s is clearly available without having a significant effect on the design.

There are a large number of possible solutions to this design problem since the value of X_n is optional provided it remains between the limits of 88.3 and 385 (s or m/s); similarly, there are a variety of choices for D and n . Table 2 summarizes a few alternatives for $H < 6$ m and $10 < L < 30$ m. The principal economic variable of the project is pump cost, which is proportional to $D^{1.35}$ (4). Since the designed pump size is proportional to X_n (see

Table 3 Other design examples

Example	q (l/min)	h (m)	H_s (m)	$h/H_s^{1/2}$ (m ^{1/2})	X_n (s, m/s)	D (mm)	H (m)	n (min ⁻¹)	L (m)	Q (l/min)	u_c (m/s)	F_c
1	2	20	0.25	40	60	25	3.2	35	16.5	25.0	1.67	1.06
2					80		4.3	50	20.5	18.7	1.25	0.80
3					100		5.3	65	24.6	15.0	1.00	0.64
4					120		6.4	80	28.8	12.5	0.83	0.53
5	12	50	0.3	91	110	76	3.8	35	14.4	315	2.27	1.32
6					150		5.2	50	18.7	231	1.67	0.97
7					200		6.9	65	25.6	173	1.25	0.73
8					230		8.0	80	27.5	151	1.09	0.63
9	20	80	0.4	126	150	102	4.8	35	15.4	666	2.67	1.35
10					200		6.4	50	19.2	499	2.00	1.01
11					250		8.0	65	23.1	400	1.60	0.81
12					300		9.6	80	27.0	333	1.33	0.67
13	50	120	0.5	170	220	250	2.9	35	9.2	4091	2.73	1.23
14					270		3.6	50	9.7	3333	2.22	1.00
15					320		4.3	65	10.5	2812	1.88	0.85
16					380		5.1	80	12.0	2368	1.58	0.71

Table 2), the smallest value of X_n consistent with topographical constraints on-site will lead to the most cost effective design solution. It should also be noted from Table 2 that, for a given pump duty, the required capacity of the source Q is inversely proportional to the supply head H .

The second design example concerns an actual project undertaken by the University's Community Development Committee in October, 1996. The site is near the former gold-mining town of Wau in Papua New Guinea. A small

settlement required 2.7 l/min of water for domestic purposes. The water source is 27 m below, and a distance of 373 m from, a ferrocement receiving tank. The delivery head was assumed to be 30 m to allow for friction losses. A 53 mm Wilcox ram pump was to be used with the valve mass (0.83 kg) as supplied and $A_{disc} = 2.55 \times 10^{-3} \text{ m}^2$; thus H_s was 0.32 m and, since $h/\sqrt{H_s} = 53 \text{ m}^{1/2}$, $60 < X_n < 330 \text{ s}$ or m/s. Taking X_2 as 140 s gives $HD^2 = 0.0063 \text{ m}^3$ and, if $D = 0.053 \text{ m}$, $H = 2.24 \text{ m}$. Choosing $n = 0.83 \text{ Hz}$ (50 min^{-1})

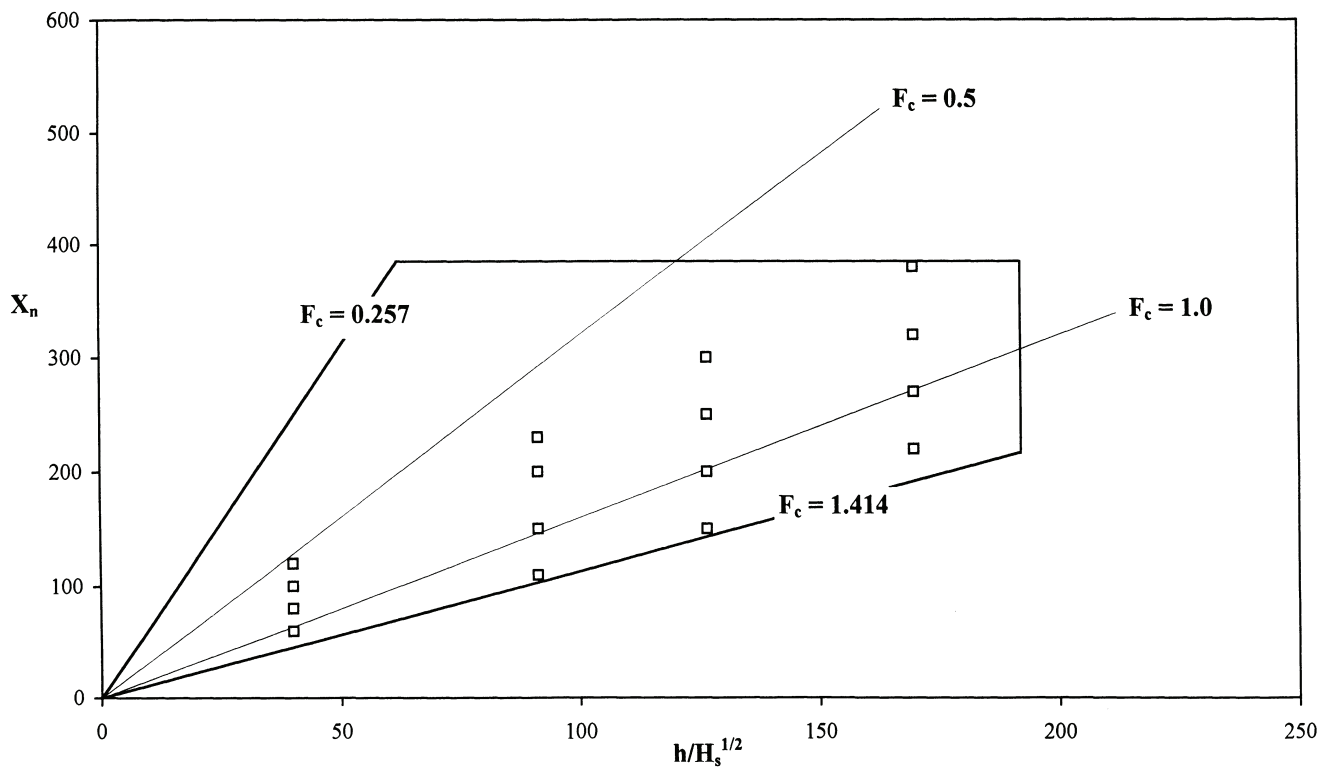


Fig. 3 Design examples plotted on the $X_n : h/\sqrt{H_s}$ diagram

with X_1 as 140 m/s gives $L = 12.5$ m. Finally, $X_3 = 140$ s and Q was determined as 1.2×10^{-3} m³/s (72 l/min). The installed system had $L = 13$ m, $H = 2.7$ m, $q = 2.7$ l/min and $Q = 78$ l/min. The beat frequency was 53 min^{-1} for a valve stroke of 8 mm. The system has been operating successfully, non-stop and without the need for any form of maintenance, for the last fifteen months (time of writing is January 1998). The total cost of the installation including the ferrocement tank, a reticulation system and three taps was in the region of £2000. There were no labour costs involved as the work was done by the villagers and British volunteers on attachment to the University through Voluntary Service Overseas.

More design examples are summarized in Table 3 and plotted on the $X_n : h/\sqrt{H_s}$ diagram in Fig. 3. Table 3 shows that for a given pump size and duty, the supply head is proportional to X_n and the source capacity is inversely proportional to both the supply head and X_n .

5 SUMMARY AND CONCLUSIONS

A generic design method has been presented which is considered to be applicable to any ram pump having a galvanized iron drive pipe. The method has a rational basis and is rapid and easy to use. It employs a single design chart featuring a feasible design space bounded by values of critical Froude number between 0.26 and 1.4. Points within this space represent design solutions for which three groups of variables (the characteristic parameters) take the same numerical value. There is ample empirical support for the procedure.

The performance of a ram pump is defined by nine separate variables, six independent and three dependent. By adopting realistic upper and lower bounds for the critical velocity of the water in the drive pipe it is possible to set limits on the values of the characteristic parameters, each of which contains one of the dependent variables in

combination with either two or three independent variables. The static head necessary to keep the impulse valve closed is required for the design of a particular pump unit and manufacturers are recommended to include this information in their literature.

Apart from the reticulation system, which should be designed separately for least cost, the principal economic variables of a water supply project employing a ram pump are the costs of the pump unit and the drive pipe, although the latter has only a minor effect. The choice of a numerical value for the characteristic parameters is optional between the defined limits, but the smaller the value selected, consistent with topographical constraints, the smaller the designed pump size and hence the cheaper the pump.

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